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EFFECT OF LUBRICANT EXTREME-PRESSURE ADDITIVES ON ROLLING-ELEMENT FATIGUE LIFE

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16. Abstract

The effects of surface active additives on rolling-element fatigue life were investigated with the five-ball fatigue tester at conditions where classical subsurface initiated rolling-element fatigue is the sole mode of failure. Test balls of AISI 52100, AISI M-50, and AISI 1018 were run with an acid-treated white oil containing either 2.5 percent sulfurized terpene, 1 percent didodecyl phosphite, or 5 percent chlorinated wax. In general, it was found that the influence of surface active additives was detrimental to rolling-element fatigue life. The chlorinated-wax additive significantly reduced fatigue life by a factor of 7. The base oil with the 2.5 percent sulfurized-terpene additive can reduce fatigue life by as much as 50 percent. No statistical change in fatigue life occurred with the base oil having the 1 percent didodecyl-phosphite additive. The additives used with the base oil did not change the ranking of the bearing steels where rolling-element fatigue life was of subsurface origin.

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EFFECT OF LUBRICANT EXTREME-PRESSURE ADDITIVES ON ROLLING-ELEMENT FATIGUE LIFE

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SUMMARY

Rolling-element fatigue tests were conducted in the five-ball fatigue tester with a base oil with and without surface active additives. Three steel ball materials were investigated. The 12.7-millimeter- (0.500-in.-) diameter test balls were either AISI 52100, AISI M-50, or AISI 1018 steel. The test lubricant was an acid-treated white oil containing either 2.5 percent sulfurized terpene, 1 percent didodecyl phosphite, or 5 percent chlorinated wax. Nine combinations of materials and lubricant additives were tested at test conditions including a maximum Hertz stress of 5.52×10^{9} N/m^2 (800 000 psi), a shaft speed of 10 700 rpm, and a race temperature of 340 K (150° F).

In general, it was found that the influence of surface active additives was detrimental to rolling-element fatigue life.

The chlorinated-wax additive significantly reduced fatigue life by a factor of 7. Rolling-element surface distress was observed in some of the tests. These results suggest that the rheology of the base oil may have been altered by this additive. The base oil with the 2.5 percent sulfurized terpene additive reduced rolling-element fatigue life by as much as 50 percent. No statistical change in fatigue life occurred with the base oil having the 1 percent didodecyl-phosphite additive.

The additives used with the base oil did not change the life ranking of bearing steels in these tests where rolling-element fatigue was of subsurface origin.

INTRODUCTION

Lubricant additives can prevent or minimize wear and surface damage to bearings and gears whose components are in contact under very thin film or boundary lubrication conditions. These antiwear or extreme pressure (EP) additives either adsorb onto the surfaces or react with the surfaces to form protective coatings or surface films. The value of additives for this protection is well known, and their use is commonplace.

In rolling-element bearings, these additives aid in protecting the rubbing contacts between the cage and the balls or rollers and between the cage and the race guiding lands. Also, it has been shown (ref. 1) that antiwear additives can protect the ball-race contacts in ball bearings operating under high-temperature, high-speed conditions where lubrication conditions are marginal. In the tests of reference 1, gross surface distress and wear were eliminated with the addition of a low concentration of a substituted organic phosphonate antiwear additive to a synthetic paraffinic lubricant. Further work (ref. 2) confirmed that significant surface films are generated under similar high-temperature, high-speed conditions. These surface films produced an apparent increase in the elastohydrodynamic (EHD) film thickness between rolling disks as measured by the X-ray transmission technique.

The effects of lubricant antiwear and EP additives on rolling-element fatigue life are not well defined. When a rolling-element bearing is operating under conditions with a full EHD film separating the rolling elements, little, if any, asperity contact occurs and the life of the bearing is limited only by rolling-element fatigue. If, however, the film thickness is reduced such that significant asperity contact occurs, the life of the bearing is reduced (refs. 3 and 4). This life reduction is greater with increased frequency of asperity contact. Under these conditions, classical subsurface initiated rolling-element fatigue becomes a less common mode of failure, and excessive surface distress and smearing can become the predominant mode of failure (ref. 5). Under these extreme conditions, surface active additives may be expected to influence bearing life by preventing some of the surface damage as was demonstrated in reference 1. However, under full EHD conditions where subsurface initiated rolling-element fatigue is the criterion of failure, these surface active additives should have no effect on bearing life unless the lubricant rheology is significantly altered by the additive or the presence of the surface films.

In reference 6, the effects of several surface active additives on several bearing steels were investigated in a rolling four-ball tester. Here it was found that either beneficial or detrimental effects on life are obtained with each of several additives depending on the choice of steel. These tests (ref. 6) were run under very severe lubricant film conditions where significant surface film effects would be expected. The limiting life in the majority of these tests was of a surface distress type and not classical rolling-element fatigue.

In order to determine the effects of antiwear and EP additives on rolling-element fatigue life, it is necessary to run fatigue tests at EHD film conditions where classical subsurface rolling-element fatigue is the sole mode of failure. This objective was accomplished by testing in the NASA five-ball fatigue tester at test conditions where classical subsurface rolling-element fatigue can be expected. The significance of these tests was enhanced by the cooperation of the author of reference 6 through whom the lubricants, additives, and test balls, which were from the same batches of material as used in reference 6, were obtained.

Test conditions in the five-ball fatigue tester included a maximum Hertz stress of 5.52×10^{9} N/m^{2} (800 000 psi), a shaft speed of 10 700 rpm, a contact angle of 30°, and a race temperature of 340 K (150° F). The test balls were 12.7-millimeter- (0.500-in.-) diameter AISI 52100, AISI M-50, and AISI 1018 steel. The test lubricant was an acid-treated white oil containing an oxidation inhibitor. Tests were run with this oil without antiwear additives and with a low concentration of one of three antiwear additives.

APPARATUS AND PROCEDURE

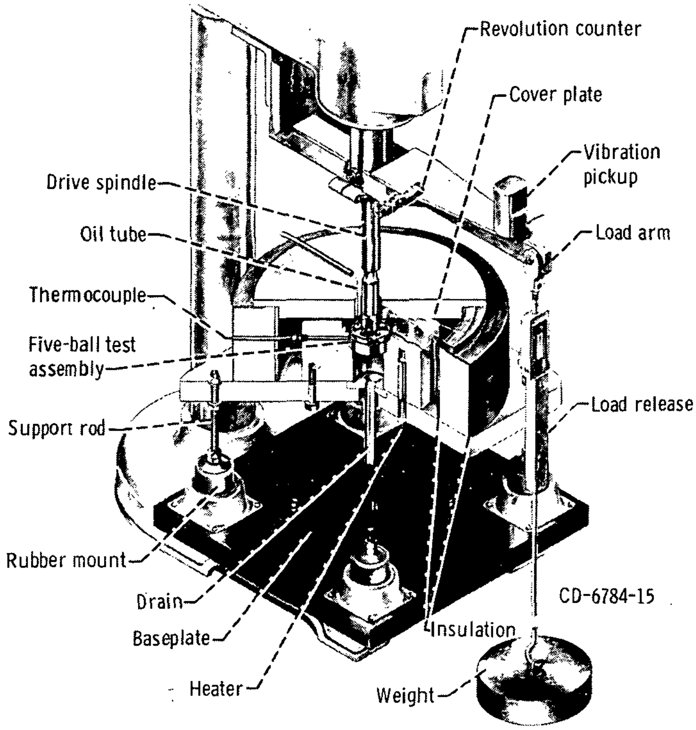
Five-Ball Fatigue Tester

The NASA five-ball fatigue tester was used for all tests conducted. The apparatus is shown schematically in figure 1 and has been described in detail in reference 7. This fatigue tester consists essentially of an upper-test ball pyramided upon four lower test balls that are positioned by a separator and are free to rotate in an angular-contact raceway. System loading and drive are supplied through a vertical drive shaft, which grips the upper test ball. For every revolution of the drive shaft, the upper test ball received three stress cycles from the lower test balls. The upper test ball and raceway are analogous in operation to the inner and outer races of a bearing, respectively. The separator and the lower balls function in a manner similar to the cage and the balls in a bearing.

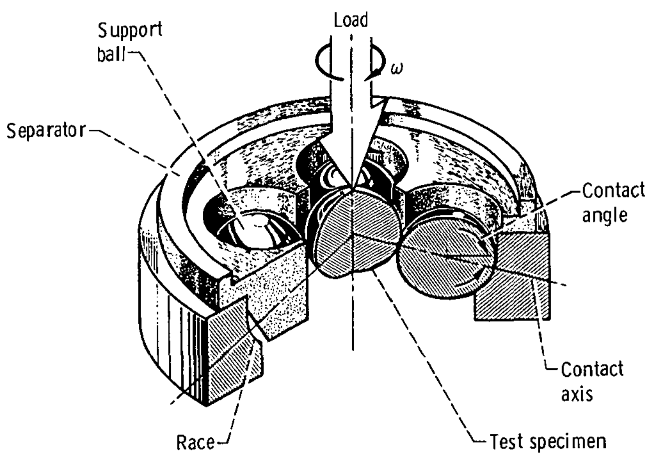
Lubrication is provided by a once-through, mist lubrication system. Vibration instrumentation detects a fatigue failure on either the upper or a lower test ball and automatically shuts down the tester. This provision allows unattended operation and a consistent criterion for failure.

Lubricants

The base lubricant used in this study was an acid-treated white oil containing 0.4 percent of 2,6 ditertiary butyl-4-methyl phenol as an oxidation inhibitor. This oil has a kinematic viscosity of 10.3×10^{-6} m^{2}/sec (10.3 cS) at 311 K (100° F) and 2.45×10^{-6} m^{2}/sec (2.45 cS) at 372 K (210° F). The hydrocarbon composition was 49 percent paraffinic, 49 percent naphthenic, and 2 percent aromatic as determined by the n-d-M method (ref. 8). Each of three additives was studied in the following concentrations in the base oil: 2.5 percent sulfurized terpene, 5 percent chlorinated wax, and



(a) Cutaway view of five-ball fatigue tester.



(b) Five-ball test assembly. CD-6838-15

Figure 1. - Test apparatus.

1 percent didodecyl phosphite. The water content for the lubricant additive combinations ranged from 19 to 25 ppm. The lubricants and additives were from the same batches as used in reference 6.

Materials

The 12.7-millimeter- (0.500-in.-) diameter balls used were made from AISI M-50, AISI 52100, and AISI 1018 steels. The balls were from the same lot of material as those tested in reference 6. Table I lists the properties and compositions of the test ball materials. The 52100 material was carbon vacuum deoxidized (CVD) whereas the M-50 and 1018 were commercially available products of unidentified melting process. The 52100 and M-50 were through-hardened whereas the 1018 was case-carburized.

TABLE I. - PROPERTIES AND COMPOSITION OF

TEST BALL MATERIALS

Property	Material		
	AISI 52100	AISI M-50	AISI 1018
Hardness, R_c	63.5	64 to 65	[a]63 to 64
Retained austenite at surface, percent	26	0	29
Composition, weight percent of element			
C	1.14	0.80	0.25
Mn	.35	.47	.78
Si	.29	.18	.01
Ni	.10	.30	.03
Cr	1.43	4.18	----
Mo	.005	3.50	.01
V	.01	1.04	.002
S	------	.004	.02
P	------	.012	.01
Fe	Balance	Balance	Balance

[a]Case carburized; core hardness 33 to 36 R_c; effective case depth, 0.15 cm (0.060 in.).

Fatigue Testing

Before they were assembled in the five-ball fatigue tester, all test-section components were flushed and scrubbed with ethyl alcohol and wiped dry with clean cheesecloth. The test balls were examined for imperfections at a magnification of 15 diameters. After examination, all test balls were coated with test lubricant to prevent corrosion and wear at startup. A new set of five balls were used for each test. Each test was terminated when a fatigue failure occurred (on either an upper test ball or a lower test ball) or when a preset cutoff time was reached. The speed, outer-race temperature, and oil flow were monitored and recorded at regular intervals. After each test, the outer race of the five-ball system was examined visually for damage. If any damage was discovered, the race would be replaced before further testing. The maximum Hertz stress that was developed in the contact area was calculated by using the formulas given in reference 9. This calculation is for dry, static contact of two elastic bodies and does not consider the effects of a lubricant in the contact area.

Method of Presenting Fatigue Results

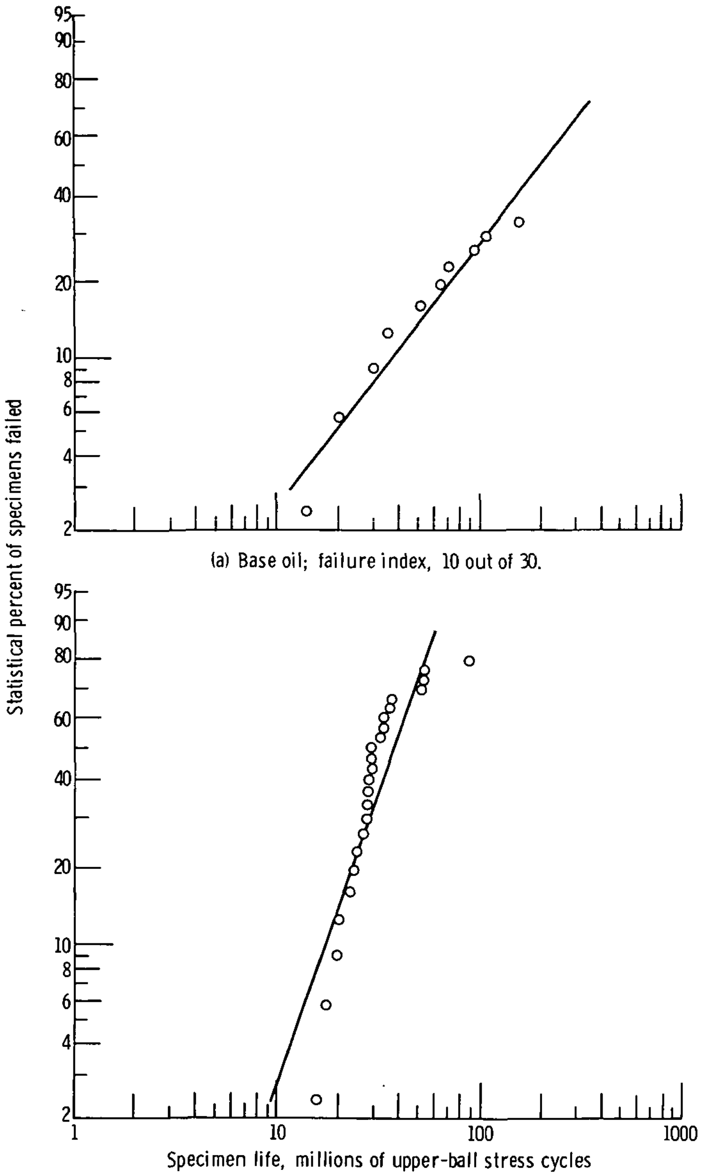
The statistical methods of reference 10 for analyzing rolling-element fatigue data were used. A plot of the log-log of the reciprocal of the probability of survival as a function of the log of upper-ball stress cycles to failure (Weibull coordinates) was obtained. For convenience, the ordinate is graduated in statistical percent of specimens failed. From a plot such as this the number of upper-ball stress cycles necessary to fail any given portion of the specimen group may be determined.

For purposes of comparison, the 10-percent life on the Weibull plot was used. The 10-percent life is the number of upper-ball stress cycles within which 10 percent of the specimens can be expected to fail; this 10-percent life is equivalent to a 90-percent probability of survival. The failure index is the number of specimens that failed out of those tested.

RESULTS AND DISCUSSION

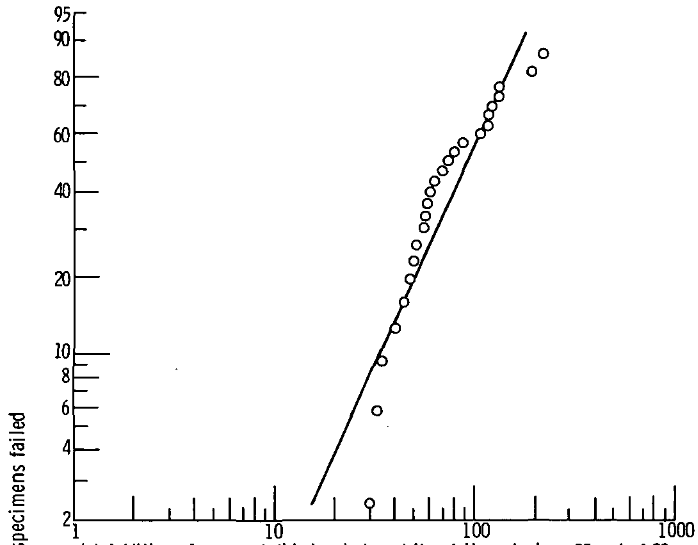
Fatigue Results

Rolling-element fatigue tests were run in the five-ball fatigue tester with a base oil with and without surface active additives with three steel ball materials. Nine combinations of materials and lubricants were tested at conditions including a maximum Hertz stress of 5.52×10^9 N/m^2 (800 000 psi), a shaft speed of 10 700 rpm, a contact angle of

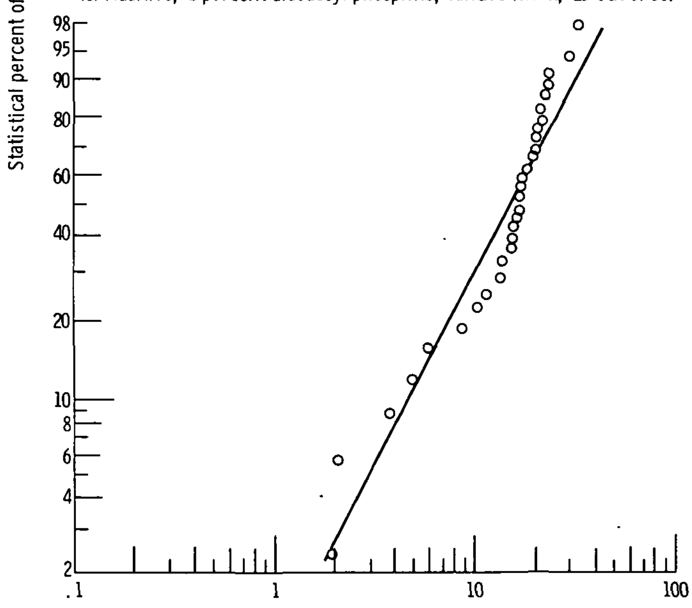


(b) Additive, 2.5 percent sulfurized terpene; failure index, 24 out of 30.

Figure 2. - Rolling-element fatigue life of AISI 52100 steel balls with an acid-treated white oil with and without additives in the five-ball fatigue tester. Maximum Hertz stress, 5.52×10^{9} N/m^2 (800 000 psi); shaft speed, 10 700 rpm; contact angle, 30°; race temperature, 340 K (150° F).

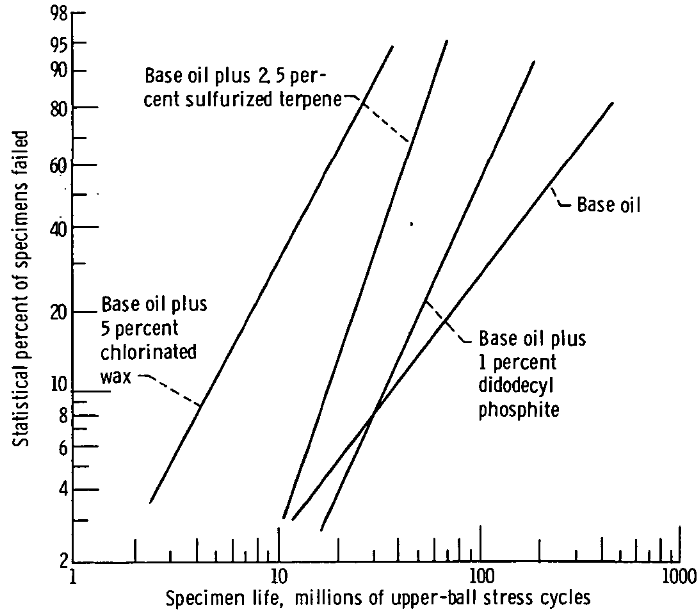


(c) Additive, 1 percent didodecyl phosphite; failure index, 25 out of 30.



(d) Additive, 5 percent chlorinated wax; failure index, 30 out of 30.

Figure 2. - Continued.



(e) Summary.

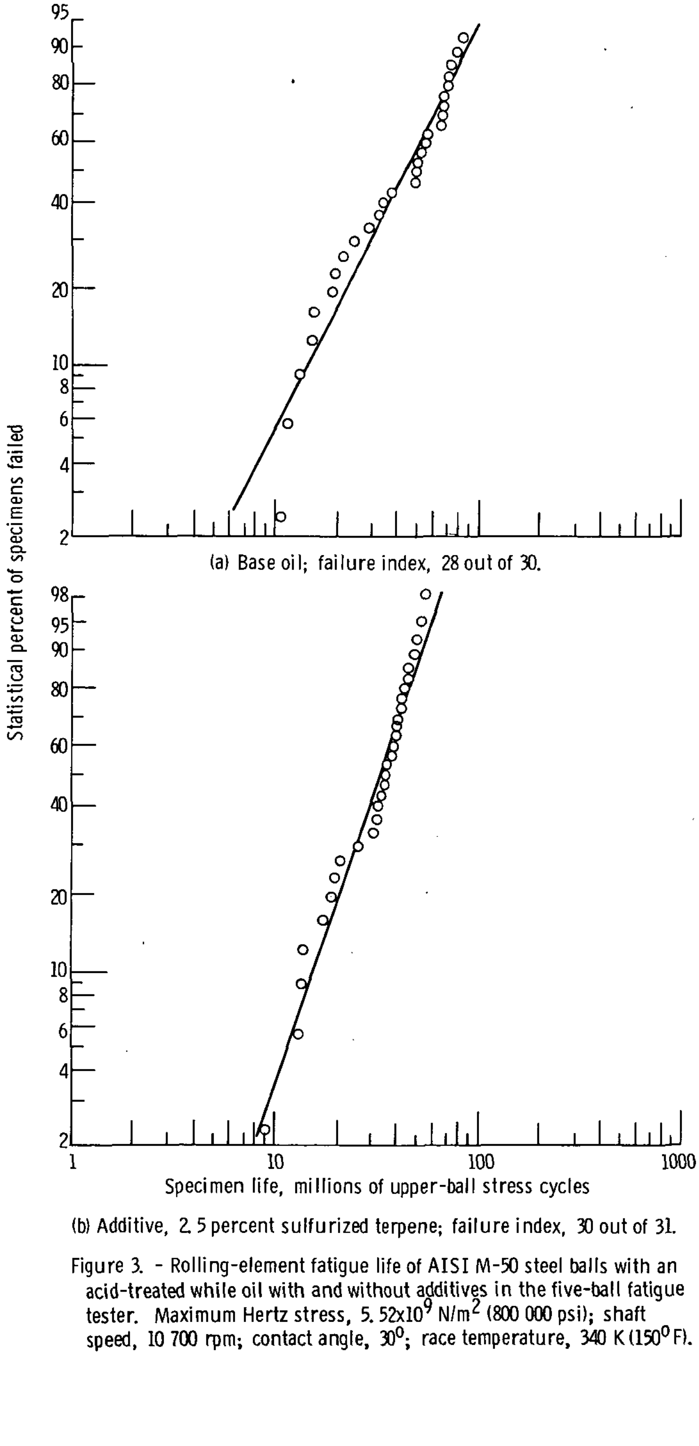
Figure 2. - Concluded.

30°, and a race temperature of 340 K (150° F). The 12.7-millimeter- (0.500-in.-) diameter test balls were either AISI 52100, AISI M-50, or AISI 1018 steel. The test lubricant was an acid-treated white oil containing either 2.5 percent sulfurized terpene, 1 percent didodecyl phosphite, or 5 percent chlorinated wax.

The results of the rolling-element fatigue tests are shown on Weibull coordinates in figures 2 to 4 and are summarized in table II. In general, for each material, the additives tended to reduce the 10-percent life. The confidence numbers calculated by methods of reference 10 and shown in table II are indicators of the significance of the additive effects on life. These confidence numbers indicate the percentage of the time that the 10-percent life with the base oil will be greater than (or less than, as the case may be) that with the additive containing oil. A confidence number of 95 percent is equivalent to a 2 sigma confidence and would indicate a significant difference in the life results in question. None of these life differences can be considered significant except the 52100-chlorinated wax combination which showed a definite life reduction.

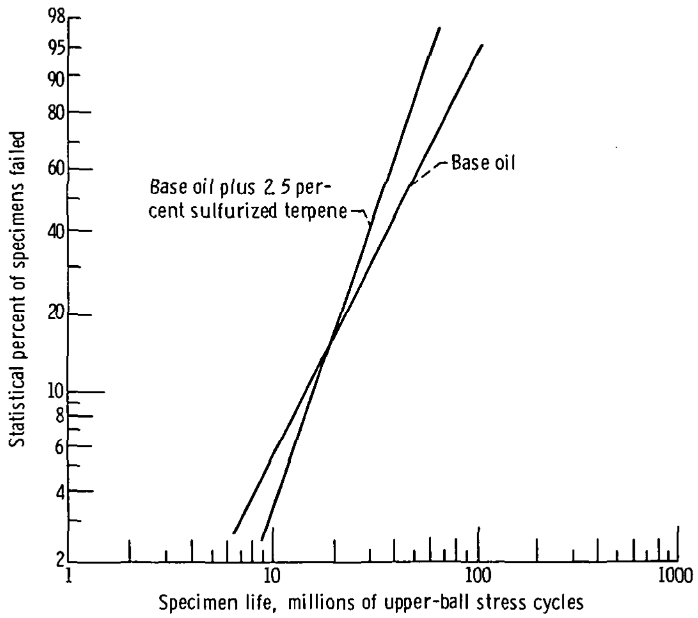
All of the fatigue failures considered in the previous analysis were classical rolling-element fatigue spalls. In other words, each failed ball had a single fatigue spall on a running track (fig. 5) which showed no surface distress that would result from asperity interaction between the ball surfaces in contact.

Figure 6 shows a comparison of the fatigue life results from this study compared with the 50-percent life data from reference 6. The results from reference 6 show both reduced and enhanced life depending on the material-lubricant additive combination.



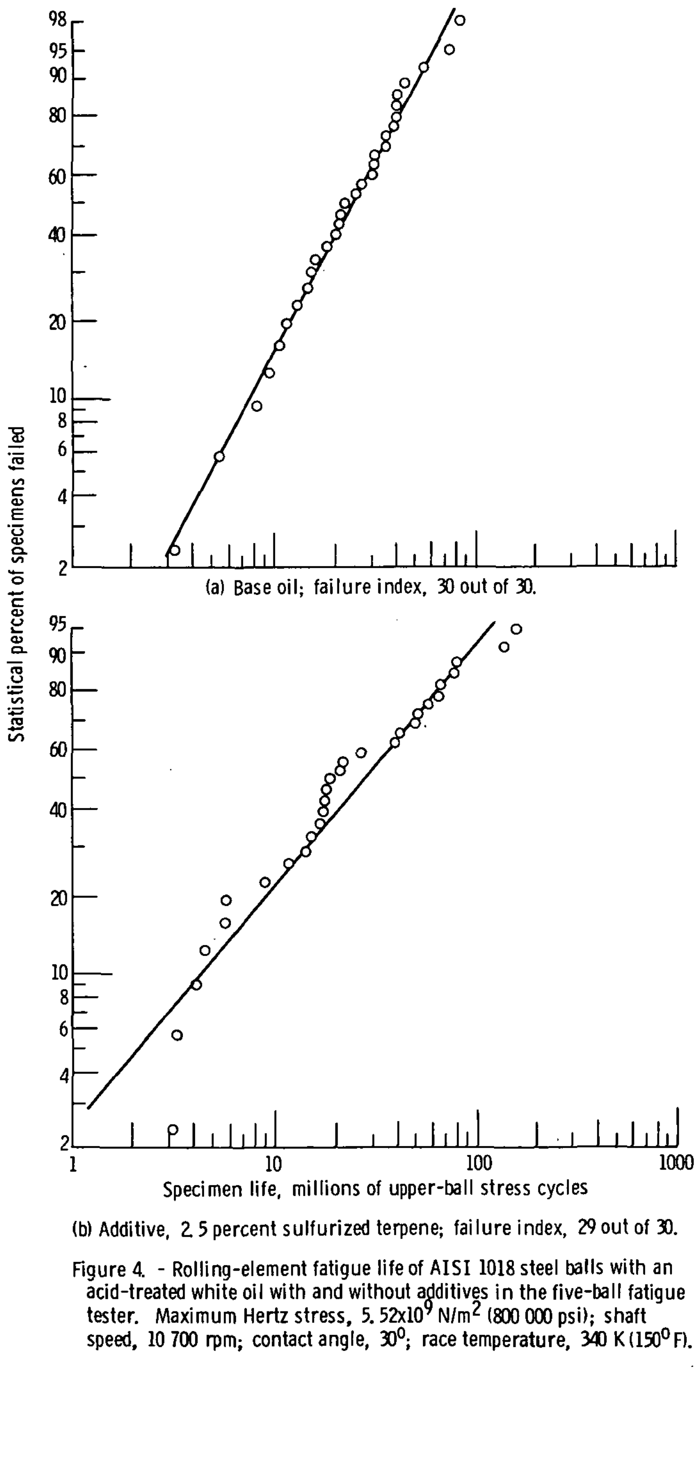
(b) Additive, 2.5 percent sulfurized terpene; failure index, 30 out of 31.

Figure 3. - Rolling-element fatigue life of AISI M-50 steel balls with an acid-treated while oil with and without additives in the five-ball fatigue tester. Maximum Hertz stress, 5.52×10^9 N/m^2 (800 000 psi); shaft speed, 10 700 rpm; contact angle, 30°; race temperature, 340 K (150° F).



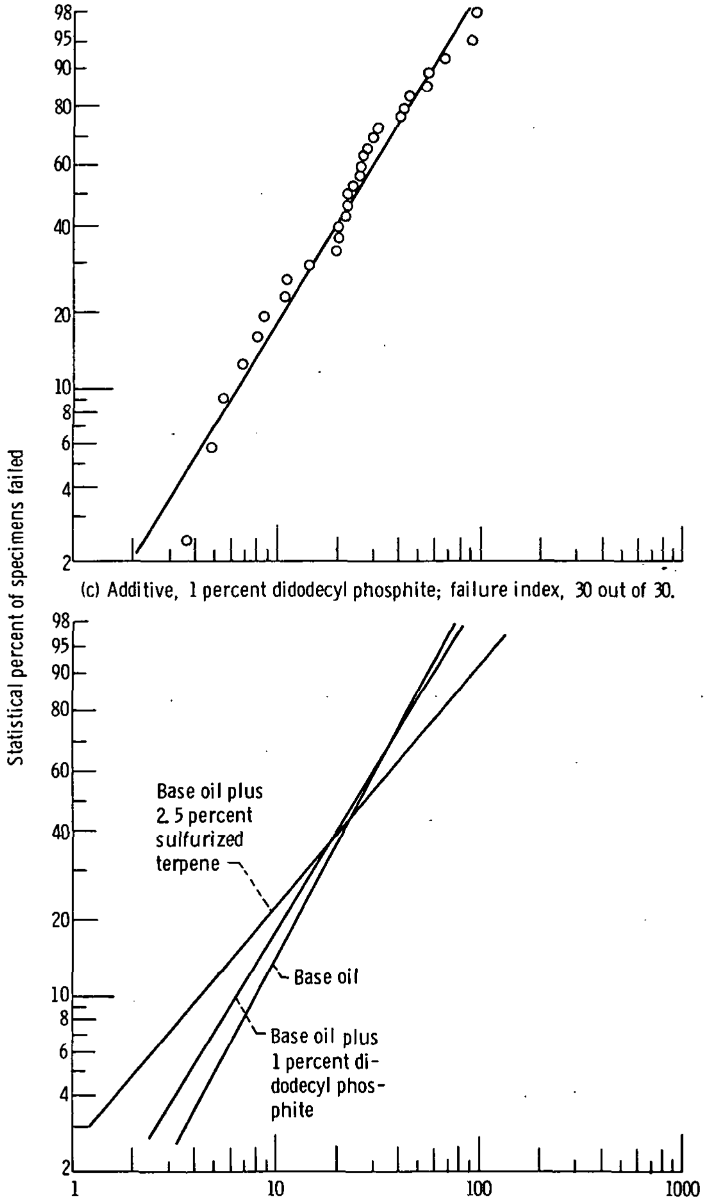
(c) Summary.

Figure 3. - Concluded.



(b) Additive, 2.5 percent sulfurized terpene; failure index, 29 out of 30.

Figure 4. - Rolling-element fatigue life of AISI 1018 steel balls with an acid-treated white oil with and without additives in the five-ball fatigue tester. Maximum Hertz stress, 5.52×10^{9} N/m^2 (800 000 psi); shaft speed, 10 700 rpm; contact angle, 30^0; race temperature, 340 K (150^0 F).



(c) Additive, 1 percent didodecyl phosphite; failure index, 30 out of 30.

(d) Summary.

Figure 4. - Concluded.

TABLE II. - ROLLING-ELEMENT FATIGUE LIFE OF STEEL BALLS WITH AN ACID-TREATED WHITE OIL WITH AND WITHOUT SURFACE-ACTIVE ADDITIVES

[Maximum Hertz stress, 5.52×10^9 N/m^2 (800 000 psi); shaft speed, 10 700 rpm; contact angle, 30°; race temperature, 340 K (150° F).]

Material	Lubricant	Rolling-element fatigue life, millions of upper-ball stress cycles		Weibull slope	Failure index	Confidence number, percent
		L_{10}	L_{50}		(a)	(b)
AISI 52100	Base oil	37.0	202	1.11	10 out of 30	--
	Base oil plus 2.5 percent sulfurized terpene	17.8	39.2	2.39	24 out of 30	81
	Base oil plus 1 percent didodecyl phosphite	33.9	92.8	1.87	25 out of 30	54
	Base oil plus 5 percent chlorinated wax	4.8	15.1	1.63	30 out of 30	99
AISI M-50	Base oil	14.6	44.5	1.69	28 out of 30	--
	Base oil plus 2.5 percent sulfurized terpene	15.9	34.1	2.47	30 out of 31	59
AISI 1018	Base oil	7.8	25.0	1.61	30 out of 30	--
	Base oil plus 2.5 percent sulfurized terpene	4.2	26.7	1.02	29 out of 30	83
	Base oil plus 1 percent didodecyl phosphite	6.4	24.4	1.40	30 out of 30	65

[a]Indicates number of failures out of total number of tests.

[b]Percentage of time that the 10-percent life with the base oil will be greater than (or less than, as the case may be) the 10-percent life with the additive.

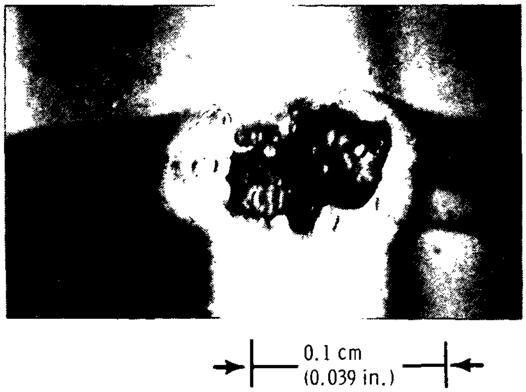


Figure 5. - Typical subsurface originated rolling-element fatigue spalls on upper test balls in five-ball fatigue tester.

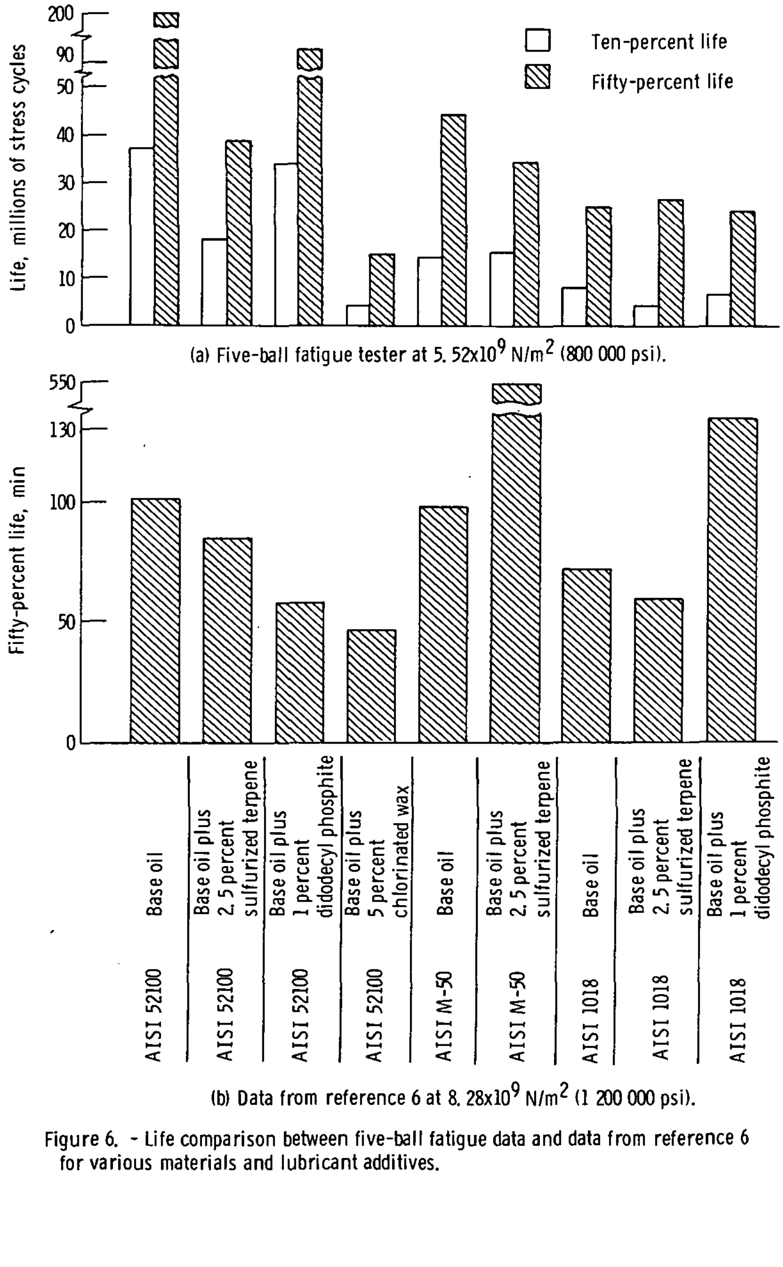


Figure 6. - Life comparison between five-ball fatigue data and data from reference 6 for various materials and lubricant additives.

Effects on Bearing Steel

The data of reference 6 indicates that lubricant additive effects can radically change the life ranking of bearing steels. In reference 6, as an example, the base oil without the additives ranked the bearing steels in the following order of decreasing life: (1) AISI 52100, (2) AISI M-50, and (3) AISI 1018. The base oil plus 2.5 percent sulfurized terpene ranked the materials as follows: (1) AISI M-50, (2) AISI 52100, and (3) AISI 1018. For the base oil plus the 1 percent didodecyl phosphite which were run with the AISI M-50 and AISI 1018 steels only, the materials were ranked as follows: (1) AISI 1018 and (2) AISI M-50. These results from reference 6 would be contrary to the conclusions of reference 11, at least with regard to the AISI M-50 and AISI 52100 which show an interrelation between rolling-element fatigue life and alloying elements for through-hardened materials. As a result, AISI 52100 should have had the longer fatigue life. Because AISI 1018 was case-carburized, no generalization can be made with regard to that material.

Examination of the data presented herein and summarized in table II reveals that the materials tested are ranked as follows regardless of additive content of the lubricant: (1) AISI 52100, (2) AISI M-50, and (3) AISI 1018. Using the methods given in reference 10, combined confidence numbers for successive test series can be calculated. The combined confidence numbers for these materials when compared to AISI 52100 are 88 percent and >99 percent for the AISI M-50 and AISI 1018, respectively.

The discrepancy between these test results and those of reference 6 can be explained in part by the number of failures in a test series and the test conditions. Since the number of failures in the test series of reference 6 was generally less than those reported herein, the differences in life (with one exception) are probably not statistically significant. It therefore may be concluded that the additives used with the base oil did not change the ranking of bearing steels where rolling-element fatigue was of subsurface origin.

Additive Effect

Base oil plus 2.5 percent sulfurized terpene. - For both the AISI 52100 and AISI 1018 steels the 2.5 percent sulfurized-terpene additive apparently reduced fatigue life approximately 50 percent. For the AISI M-50, however, the fatigue life was essentially unchanged from that obtained with the base oil. The combined confidence number for the three successive tests is 92 percent. Combining the data for the three materials assumes that the reduction in life is unrelated to the bearing steel chemistry. With this assumption, it can be concluded that the 2.5 percent sulfurized terpene can reduce rolling-element fatigue life by as much as 50 percent under the test conditions reported.

Base oil plus 1 percent didodecyl phosphite. - The 1 percent didodecyl-phosphite additive was run with the AISI 52100 and AISI 1018. The combined confidence number for these two series of tests was 68 percent or approximately equivalent to a 1 sigma confidence and was not considered statistically significant. The results with this additive showed essentially no statistical difference between the base oil with or without the additive.

Base oil plus 5 percent chlorinated wax. - Only in the tests with the chlorinated-wax additive with AISI 52100 balls was surface distress observed. In this case, eight tests were not included in the analysis because of multiple spalling and considerable surface distress in the running track. Even with these early failures deleted from the analysis, the life with this additive was significantly less than that without the additive.

It was evident that with this lubricant-additive-material-test condition, the EHD film was marginal. This result suggests the possibility that the lubricant rheology of the base oil (viscosity and/or pressure-viscosity effects) has been altered by the chlorinated-wax additive. Further, the influence of this additive can be detrimental to rolling-element fatigue life.

Elastohydrodynamic Film Effects

The failures in the majority of the tests of reference 6 were of a surface distress type rather than classical rolling-element fatigue. The operating conditions for these tests were such as to provide marginal EHD films. Under these conditions, surface films (either adsorbed or reaction films) could have an effect on life as limited by the surface distress type failure.

As a comparison of the lubricant EHD film conditions for the present five-ball fatigue tests and the four-ball tests of reference 6, the relative EHD film thickness for the two test conditions were calculated (see the appendix). This comparison shows that the theoretical film thickness in the tests of reference 6 may be nearly an order of magnitude less than that in the five-ball fatigue tests. The accuracy of the EHD theory at these high stresses has not been demonstrated. However, the results of this comparison tend to confirm the observed differences in lubricant film conditions between the two sets of tests.

Under high contact stresses and thin EHD films, the additives may affect the lubricant rheology sufficiently to reduce the EHD film thickness. It is well known that rolling-element fatigue life decreases as the ratio of EHD film thickness to composite surface roughness decreases (refs. 3, 4, and 12). The composite surface roughness σ is defined as

$$\sigma = \left(\sigma_1^2 + \sigma_2^2\right)^{1/2}$$

where σ_1 and σ_2 are the surface roughness of each of the contacting surfaces. As a result, it is not unreasonable that statistically significant life differences can occur. The question remains, however, at lower stress levels where actual bearings operate, that is, less than 2.4×10^9 N/m^2 (350 000 psi) maximum Hertz stress, can the same effects of additives be anticipated? The research reported herein does not provide an answer to this question. However, if these effects can be extrapolated to these lower stress levels, additive selection takes on a new dimension in determining rolling-element bearing reliability and life.

SUMMARY OF RESULTS

Rolling-element fatigue tests were run in the five-ball fatigue tester with a base oil with and without surface active additives. Three steel ball materials were investigated. The 12.7-millimeter- (0.500-in.-) diameter test balls were either AISI 52100, AISI M-50, or AISI 1018 steel. Test conditions included a maximum Hertz stress of 5.52×10^9 N/m^2 (800 000 psi), a shaft speed of 10 700 rpm, a contact angle of 30°, and a race temperature of 340 K (150° F). The test lubricant was an acid-treated white oil containing either 2.5 percent sulfurized terpene, 1 percent didodecyl phosphite, or 5 percent chlorinated wax. The following results were obtained:

1. The influence of surface active additives can be detrimental to rolling-element fatigue life.

2. The base oil with the chlorinated-wax additive significantly reduced fatigue life by a factor of 7. Rolling-element surface distress was observed in some of the tests. These results suggest that the rheology of the base oil may have been altered by this additive.

3. The base oil with the 2.5 percent sulfurized-terpene additive can reduce rolling-element fatigue life by as much as 50 percent under the test conditions reported.

4. No statistically significant change in fatigue life occurred with the base oil having the 1 percent didodecyl-phosphite additive.

5. The additives did not change the life ranking of bearing steels in these tests where rolling-element fatigue was of subsurface origin.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, June 1, 1973,
501-24.

APPENDIX - ELASTOHYDRODYNAMIC FILM THICKNESS ANALYSIS

The difference in the severity of the test conditions between the five-ball fatigue tests reported herein and the four-ball tests of reference 6 can be estimated by comparing the elastohydrodynamic film thickness at the two conditions. The significant differences in test conditions were shaft speed, maximum Hertz stress, and temperature, all of which affect EHD film thickness according to the following relation from reference 13. For point contact (a ball on a ball)

$$h \propto (\mu_o u)^{0.725} P_{Hz}^{-0.174}$$

where

h minimum film thickness

μ_o lubricant ambient absolute viscosity

u rolling velocity

P_{Hz} maximum Hertz stress

If the subscript 1 denotes the conditions of reference 6 and subscript 2 denotes the five-ball fatigue tester conditions, then

$$\frac{h_1}{h_2} = \left(\frac{\mu_{o_1} u_1}{\mu_{o_2} u_2}\right)^{0.725} \left(\frac{P_{Hz_2}}{P_{Hz_1}}\right)^{0.174}$$

and

$$\frac{u_1}{u_2} \approx \frac{1580 \text{ rpm}}{10\ 700 \text{ rpm}} = 0.15$$

Since the ball diameter is the same in both tests. Also,

$$\frac{P_{Hz_2}}{P_{Hz_1}} = \frac{5.52\times10^9 \text{ N/m}^2}{8.28\times10^9 \text{ N/m}^2} = 0.67$$

and $T_2 = 340$ K (150° F) and $T_1 = 366$ K (200° F) so that

$$\frac{\mu_{o_1}}{\mu_{o_2}} \approx \frac{2.7\ \mathrm{cP}}{4.7\ \mathrm{cP}} = 0.57$$

Then

$$\frac{h_1}{h_2} = (0.57 \times 0.15)^{0.725}(0.67)^{0.174} = 0.16$$

Therefore, the theoretical film thickness at the more severe test conditions of reference 6 is nearly an order of magnitude less than that for the present five-ball fatigue tests. The accuracy of EHD theory at these very high stress levels may be questionable. However, this analysis does provide an indication of the relative severity of the two test conditions.

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